The importance of transient data analysis

by Richard Thomas
Senior Engineer
Machinery Diagnostic Services
Bently Nevada Corporation
Baton Rouge, Louisiana

Transient data reveals much about a machine's condition that steady state data cannot. New technologies make this information much more available for evaluating the mechanical integrity of a machine train. Often, though, machinery audits do not use transient data. Recent surveys indicate that throughout industry, simplicity is preferred over completeness in machine audits. This is most clearly shown by the number of rotating machinery predictive maintenance programs in which only static, steady state vibration amplitude is trended, while important transient vibration data is not even acquired.

Consider the analogy of an airplane pilot. He must be aware of:

- Instantaneous speed and altitude (amplitude).
- Instantaneous direction (phase).
- Estimated time of arrival (amplitude trend).
- Heading (phase trend).
- Vertical speed, ascending or descending (transient operation).

If the pilot only "trended" his speed, and ignored his direction and his rate of ascent or descent, disaster would surely follow. However, in the past decade, predictive maintenance programs increasingly emphasize "speed of flight" analysis, while de-emphasizing other, equally important data.

This article will explain what transient data is, why it's important, and the types of plots that are used for its display.

What transient data is and why it's important

Transient data is amplitude, phase, frequency, position and process data. Transient data is acquired from a machine during startup or shutdown, or during a change in speed or load.

Vibration data can be conveyed in two forms: Static and dynamic. Static data is discrete values that describe the vibration signal. For instance, a vibration signal might be described by the static values of amplitude, phase lag angle and frequency. Dynamic data is the actual vibration signal.

As a rotating machine changes speed, the forces and stiffnesses that act upon it change. Although many elements contribute to these forces and stiffnesses, a machine's vibration response is simply force divided by stiffness. A machine's vibration response, as it changes speed, tells us much about the nature of the forces and stiffness acting on it. This perspective cannot be supplied by steady state data. Transient data can identify:

- *Slow roll speed:* The maximum speed at which no dynamic motion can be measured. Shaft bow and runout can be measured at slow roll speeds.
- *Slow roll vector:* The slow roll vector is data unrelated to a machine's dynamic motion. It can mask dynamic motion data, so we measure it for use in creating compensated plots.
- *Mode shapes:* Lateral mode shape information is valuable for balanc-

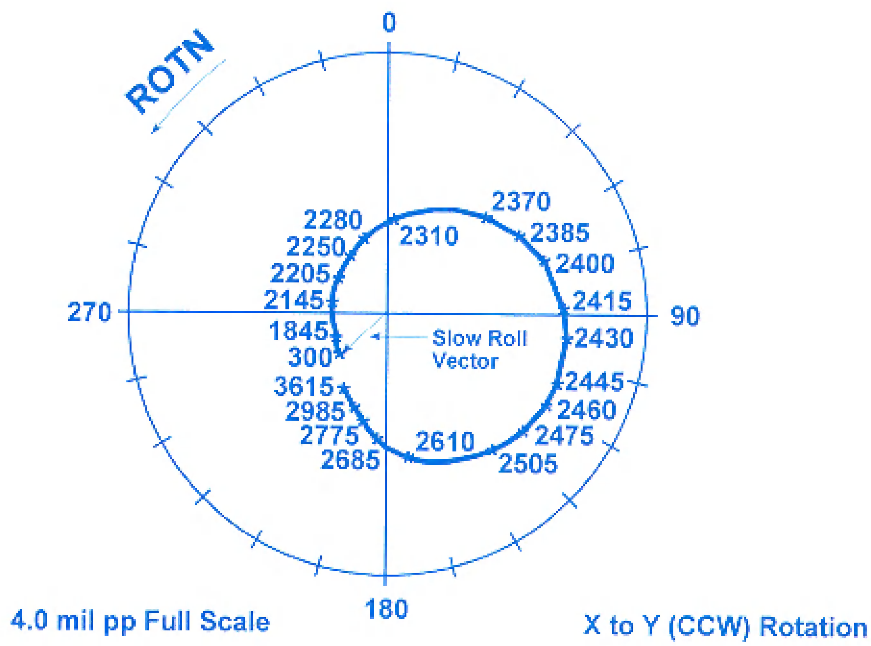

Figure 1
Uncompensated 1X polar plot

ing and for identifying faults, such as shaft cracks, bearing failures and rotor to stator rubs.

- *Heavy spot locations:* Essential information for balancing.
- *Balance resonance speeds:* Shaft rotative speeds that equal lateral natural frequencies of the rotor system. At or near these speeds, vibration amplitudes are highest. To avoid damage, a machine should not be operated at or near resonance speeds.
- *Synchronous amplification factor:* A measure of the rotor's susceptibility to vibration when rotational speed is equal to a rotor lateral natural frequency.
- *Synchronous quadrature dynamic stiffness:* Inversely related to the synchronous amplification factor, it is an indicator of rotor system damping. This damping acts as a "dam" against forward circular whip type malfunctions.
- *Load:* A unidirectional steady state force acting on the rotor system. Misalignment caused by a load is one of the largest contributors to machine failure and reduced lifespan.
- *Rubs:* Contact between a machine's rotating and stationary parts.
- *Instabilities:* The most common are the fluid-induced instabilities, whirl and whip. Instabilities can cause destructive levels of vibration.
- *Shaft cracks:* A cracked shaft can cause the most catastrophic failures.

Transient data can be acquired from both machine startups and shutdowns. Data acquired from startups and shutdowns may differ slightly because of differences in driving torque and in the thermal and alignment states of the machine.

Transient data formats

The most common presentation plots used for transient data analysis are:

- Polar
- Bode
- Shaft average centerline
- Orbit
- Spectrum cascade

Polar and Bode plots

Polar and Bode plots both show the change in a filtered vibration vector with changes in shaft rotational speed. On a polar plot, the two vector components, phase and amplitude, are plotted on the same graph, in polar format. On a Bode plot, phase and amplitude are plotted on separate XY (rectangular) graphs. Each plot format makes certain vibration characteristics easier to identify. They complement each other; therefore, you should always view polar and Bode plots together.

A polar plot (Figures 1 and 2) is a polar format presentation of the locus of the shaft 1X (or 2X,...) filtered vibration vector from a single channel as a function of shaft rotative speed, usually acquired during machine startup or coastdown (transient operation).

A Bode plot (Figures 3 and 4) is a pair of graphs in XY format displaying the 1X (or 2X, 3X ...) vibration vector from a single channel as a function of shaft rotative speed. The Y axis of the top graph represents phase lag angle, while the Y axis of the bottom graph represents amplitude. The common X axis represents shaft rotative speed. A 1X Bode plot is sometimes called an unbalance response plot.

Usually, we plot 1X data in these formats, although we can also plot the harmonics of running speed (2X, 3X,...). Higher-order transient data plots, not discussed in this article, are valid and useful tools.

Polar and Bode plots identify the slow roll runout vector, the slow roll speed limit, the balance resonance speeds, the synchronous amplification factor, the synchronous quadrature dynamic stiffness and the location of heavy spots. Polar and Bode plots are essential for identifying changes in resonance frequency due to malfunction mechanisms, for instance, rubs and cracked shafts.

The slow roll vector and slow roll speed range

The slow roll vector is data unrelated to a machine's dynamic motion. It is caused by rotor bow, mechanical runout and electrical runout. We compensate polar and Bode plots for slow roll runout because it can mask data that shows the machine's dynamic motion.

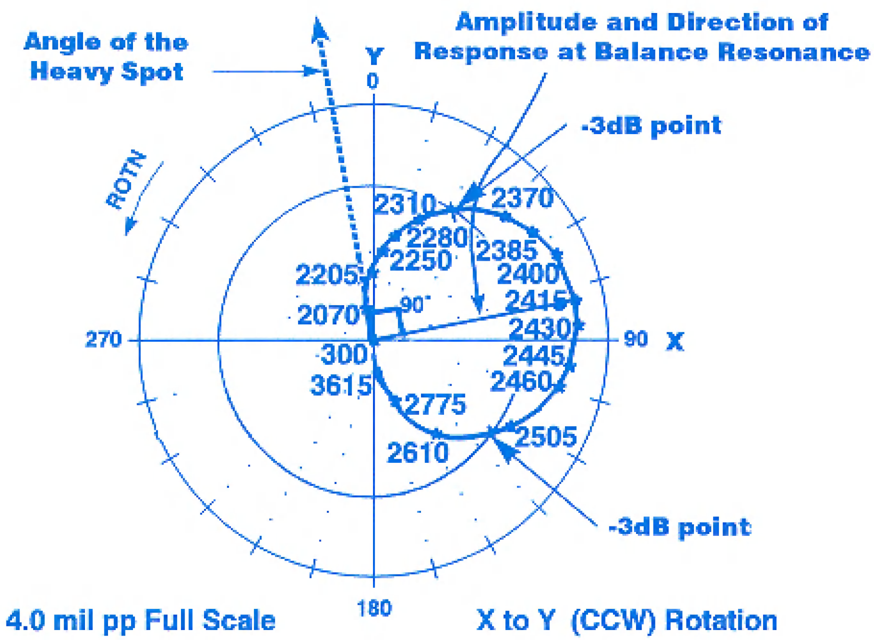

Figure 2
Compensated 1X polar plot

Most of the polar and Bode plots we use in machinery diagnostics have been compensated for slow roll runout. Usually, our only use of uncompensated plots is to identify the slow roll vector and the slow roll speed range limit.

The slow roll speed limit is the point, at very low rotational speeds, where vibration amplitude or phase begins to change. Above that speed, the machine begins to show shaft dynamic motion. Therefore, slow roll data should be acquired below this speed.

On an uncompensated Bode plot, slow roll vector components are read from the vertical axes on the phase and amplitude graphs. The values of the leftmost, flattest portion of each curve are the slow roll vector's phase and amplitude components. In the example shown in Figure 3, the 1X slow roll vector is 1.0 mils pp at 225 degrees phase lag.

On an uncompensated Bode plot, the slow roll speed limit is determined by noting the speed where either vibration phase or amplitude begins to change. Draw a vertical line from that point to the horizontal speed axis. That speed is the highest speed at which slow roll data can be acquired. In the example shown in Figure 3, slow roll data must be acquired below 1000 rpm.

On an uncompensated polar plot, read the slow roll vector by drawing a line from the origin to the lowest speed sample on the graph. The vector's amplitude is the line's length measured on the polar plot's concentric amplitude scale. The vector's phase lag component is the point on the polar plot's phase scale, measured against the direction of shaft rotation, where that line intersects if extended. In the example shown in Figure 1, the 1X slow roll vector is 1.0 mils pp at 225 degrees phase lag.

On an uncompensated polar plot, the slow roll speed range is more difficult to identify, because it has no speed scale or axis. We prefer to use the Bode plot for this measurement.

Balance resonance speeds

A balance resonance speed is a shaft rotative speed, or speed region, which equals a natural frequency of the rotor system. As shaft rotative speed approaches a rotor system natural frequency, vibration amplitude increases and vibration phase begins to lag (move against the direction of rotation) the unbalance force that causes it. At the resonance peak, the amplitude of the vibration is maximum and its phase lag is approximately 90°. As rotative speed increases past the resonance peak, the phase lag increases until it reaches approximately 180° at speeds above the resonance. At resonance, the stiffnesses that restrain shaft motion are at a minimum (as explained later for the synchronous amplification factor), so vibration amplitude is at a maximum and phase lag is changing most quickly. Therefore, a machine should never be operated at or near a balance resonance.

We use compensated polar and Bode plots to identify resonance speeds. On both plots, balance resonance speeds are identified by the characteristics that define them: a peak in amplitude accompanied by a 90° phase lag. On a Bode plot, as the amplitude plot's graph line rises to its peak, the phase plot's graph line falls by 90°.

On a polar plot, the amplitude and accompanying phase shift appear as a circle. The amplitude peak is the point on this circle farthest from the polar plot's origin. Its 90° phase lag is shown by the right angle between a vector at resonance and a vector at a speed well below resonance. First, draw a line from the origin through a point on the response circle farthest from the origin. Next, draw a line from the origin through a point on the circle at a speed well below resonance (very near the origin). The angle between the two lines is approximately 90°.

The synchronous amplification factor and the synchronous quadrature dynamic stiffness

Two other, related machinery parameters can be analyzed from polar and Bode plots. One is the synchronous amplification factor, or Q_S. Q_S is useful for estimating the vibration amplitudes that occur at rotor system natural frequencies due to the unbalance force. Inversely related to it is the synchronous quadrature dynamic stiffness.

System dynamic stiffness limits a rotating machine's vibration response, because vibration response is equal to input force(s) divided by dynamic stiffness. Dynamic stiffness is composed of two terms, a direct term and a quadrature term. The direct term produces motion in line with the force and con-

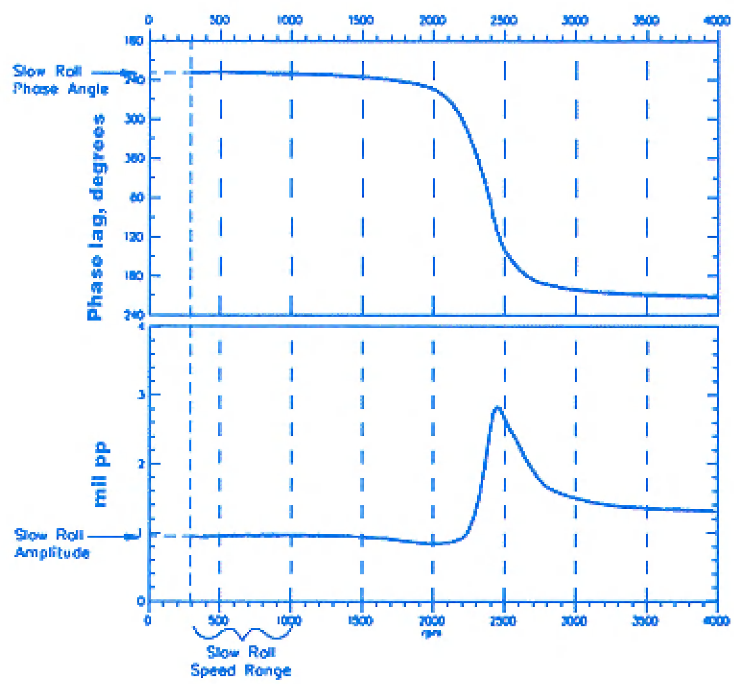

Figure 3
Uncompensated 1X Bode plot

tains a spring stiffness and a mass stiffness component. The quadrature term produces motion at 90° to the force (one *quarter* of 360°) and contains the damping stiffness terms. At a mechanical resonance, the spring stiffness and the inertial stiffness of the direct term are equal in magnitude but opposite in direction; they cancel each other. At a mechanical resonance, the only restraint is the quadrature dynamic stiffness term. Thus, the synchronous amplification factor, Q_S, is a measure of the amount of synchronous quadrature dynamic stiffness that is present in the rotor bearing system at a mechanical resonance. A high Q_S indicates a low synchronous quadrature dynamic stiffness, and a low Q_S indicates a high synchronous quadrature dynamic stiffness.

Although several criteria can be applied to evaluate the significance of Q_S, the synchronous amplification factor should be less than 5 for a well-designed system. Q_S values between 5 and 8 indicate that the rotor system has marginal quadrature stiffness at resonance. If the rotor system is not symmetric, then this measurement *cannot* be used until the resonances are properly separated. For split resonance separations, refer to BRDRC papers. Q_S values in excess of 8 indicate that the system has low quadrature stiffness at resonance and is susceptible to large amplitude excursions at resonance. High quadrature stiffness is desired because it limits vibration amplitude at resonance.

Q_S may be calculated in several ways. One is the Half-Power Bandwidth method (See *Half-Power Bandwidth method for the evaluation of synchronous and nonsynchronous quadrature stiffnesses* in the June, 1994 issue of the Orbit). To calculate Q_S by the Half-Power Bandwidth method, divide the frequency at the resonance peak by the resonance bandwidth. The resonance bandwidth is the difference in frequency between the points on either side of the resonance peak that are 3dB below (or 0.707 times) the peak amplitude. Another method is to calculate the ratio of vibration amplitude at resonance to vibration amplitude at high rotative speed (but below the next higher mode range). The following will explain how the values required for the Half-Power Bandwidth method are obtained from both the Bode and polar plots.

On the Bode plot (Figure 4), the resonance frequency occurs at 90 degrees phase lag, which is near the amplitude peak of the angle of the high spot from the angular location of the heavy spot. To calculate the resonance bandwidth, multiply the vibration amplitude of the resonance peak by 0.707. Draw a horizontal line on the Bode plot's amplitude graph at that point. The difference between the speeds at the two points the line intersects is the resonance bandwidth. In Figure 4, the peak amplitude is 3.5 mils, and occurs at 2415 rpm. 3.5 mils X 0.707 ≈ 2.5 mils pp. The two points on the graph with these vibration levels are at approximately 2330 and 2530 rpm. The difference between them (the bandwidth) is 200 rpm. In this example, Q_s = 2415 / 200 ≈ 12.

On the compensated polar plot (Figure 2), the resonance frequency is the point on the plot circle farthest from the origin. To calculate the resonance bandwidth, multiply the peak amplitude by 0.707. This value will correspond to two points on the plot; each has a speed associated with it. The difference between those speeds is the resonance bandwidth. In the example shown in Figure 2, the peak amplitude is 3.5 mils pp, and occurs at 2415 rpm. 3.5 mils pp X 0.707 ≈ 2.5 mils pp. The two points on the graph with vibration levels of 2.5 mils pp are at 2330 and 2530 rpm. The difference between them (the bandwidth) is 200 rpm. In this example, Q_S = 2415 / 200 ≈ 12.

A Q_S value calculated from a Bode plot should agree with that calculated from a polar plot. Small differences are normal, and are due to accuracy in determining values. However, a Q_S calculated from startup data and a Q_S calculated from shutdown data may differ due to thermal, load and ramp rate differences, as well as for other reasons.

Caution should be used when applying Q_S calculated by either the Half-Power Bandwidth method or the amplitude ratio method, because of potential mechanical nonlinearities and abnormalities which can affect machinery vibration response. Further, the synchronous amplification factor does not define or describe the properties that determine rotor stability. The means of determining operational stability margin is given in references #23 & 24.

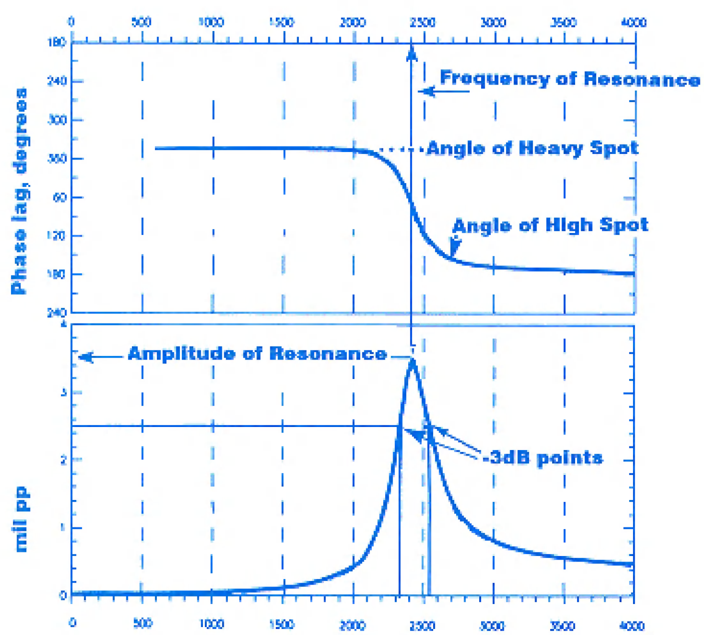

Figure 4
Compensated 1X Bode plot

Heavy spot location

A rotor that is not perfectly balanced will have a mass unbalance. The location of the mass unbalance, or heavy spot, is essential information for use in rotor balancing.

On a compensated Bode plot, the value of the leftmost, flat portion of the phase curve is the angle of the heavy spot location with respect to the transducer used to generate the plot. In the example shown in Figure 4, the angle of the heavy spot is located at 350 degrees phase lag, with respect to the transducer used to generate the plot.

On a compensated polar plot, the angle of the heavy spot location is identified by drawing a line from the origin through the lowest speed sample on the graph, to the polar plot's phase scale. In the example shown in Figure 2, the angle of the heavy spot is located at 350 degrees phase lag, with respect to the transducer used to generate the plot.

Be aware that if the probe that supplied data for the Bode or polar plot is located near a fluid-lubricated bearing or seal, the heavy spot location may not be correct - see *Precautions on Polar-plot Balancing* in the March, 1993 issue of the Orbit.

The shaft average centerline plot

The shaft average centerline plot (Figure 5) depicts the average position about which the shaft vibrates within its bearing clearance, in XY coordinates. Both transient (versus speed) and trend (versus time) data can be plotted in this format. The X and Y axes show the average change in the shaft's horizontal and vertical positions with respect to some initial or reference position.

A typical shaft average centerline plot's X and Y reference points are at the bottom center of the plot. This is because, on a typical, horizontally-mounted machine with the rotor mass between its bearings, we assume that the shaft rests in the bottom of its bearing at zero speed. Of course, if a better knowledge of shaft static position is known, that should be the starting point. The shaft average centerline plot begins at either zero or slow roll speed. As the machine is brought up to speed, the plot shows how the shaft rises on the bearing's oil wedge. The shaft average centerline plot also shows changes in the shaft's position with changes in load.

The shaft average centerline plot is useful for identifying changes in load and bearing wear, as well as for calculating the average eccentricity ratio and the rotor position angle.

Average eccentricity ratio is a relative measure of the shaft's position between the center of the bearing and the bearing wall. It is calculated by dividing the average position of the shaft centerline by the bearing (or seal) radial clearance. A shaft with a zero eccentricity ratio is concentric with the bearing or seal, while a shaft with an eccentricity ratio of 1 is in contact with the bearing or seal. A decreasing eccentricity ratio can indicate a potential stability problem.

The rotor position angle is the angle between an arbitrary reference through the center of a bearing (usually vertical down) and a line connecting the bearing and shaft centers, measured in the direction of rotation. It can indicate the presence of abnormal loads acting on a machine.

A usual shaft average centerline position of a horizontal machine has an eccentricity ratio greater than 0.6, and a rotor position angle between 20° and 50°. A smaller eccentricity ratio indicates that the shaft centerline is approaching the bearing centerline, which usually leads to a decrease in dynamic stiffness and an increase in the bearing's fluid average circumferential velocity ratio. Because the system dynamic stiffness is reduced, the rotor has a tendency to be easily excited by the dynamic and static forces which act on it. Conversely, an eccentricity ratio greater than 0.6 indicates that the shaft is approaching the bearing surface, where destabilizing forces are minimized.

The orbit plot

An orbit plot (Figure 6) shows the dynamic, two dimensional path of the centerline motion of a machine component observed by XY transducers, in the plane of those transducers. When the transducers are XY shaft proximity probes, the orbit is the shaft centerline lateral vibration, called precession. An orbit can also be displayed on an oscilloscope in the X versus Y mode. An orbit plot clearly shows:

- the maximum vibration amplitude
- the direction of maximum amplitude
- the influence of an asymmetric support stiffness
- the presence of any load forces
- frequency of vibration versus rotative speed and direction of precession (when a Keyphasor® signal is present)

The orbit is a plot of the shaft centerline's path as the shaft vibrates; therefore, many vibration characteristics and

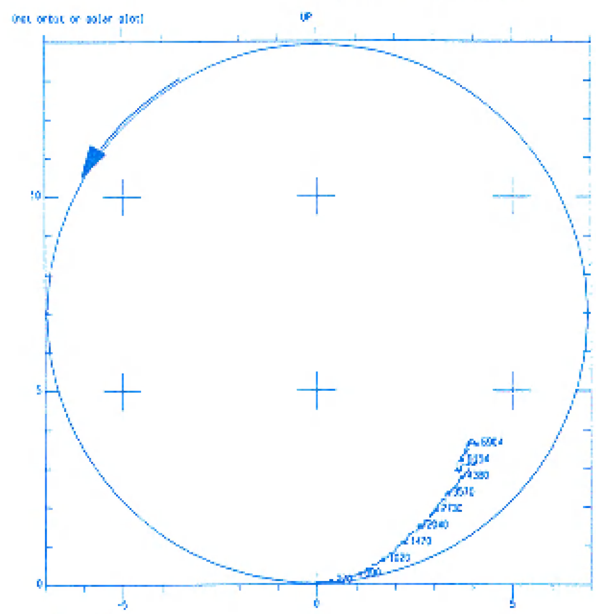

Figure 5
Shaft average centerline plot

malfunctions are easy to identify using it. The maximum vibration amplitude, and its direction, are measured on the longest axis of the orbit. Asymmetric support stiffness and unidirectional loads are identified by the shape of the orbit. If the orbit is unusually flat, or even "pinched" in one direction (Figure 6), it is indicative of one of these conditions.

Multiple orbit plots are often presented alone or in conjunction with other transient data plots to facilitate data interpretation (Figure 7).

The spectrum cascade plot

A spectrum cascade plot (Figure 7) is a graph in XY format displaying frequency spectra versus shaft rotative speeds. Shaft rotative speed and vibration amplitude are usually presented on two separate vertical (Y) axes. Frequency is displayed on the horizontal (X) axis. This data format is used to evaluate the change in vibration amplitude and frequency characteristics during machine transient conditions. The orbit plots shown in Figure 7 show the form of the vibration at various speeds. These orbit plots can be correlated to the individual spectrum plots that comprise the cascade plot.

Bently Nevada's diagnostic systems also generate an enhanced spectrum cascade plot, the full spectrum cascade plot. It displays the data from two probes, rather than one, and shows negative frequency components useful for identifying the direction of precession of individual vibration components.

Systems for transient data capture and analysis

Bently Nevada makes two excellent systems for transient data capture and analysis, Transient Data Manager® 2 and ADRE® for Windows.

The Transient Data Manager 2 (TDM2) System automatically collects and processes data during both transient and steady state operation, and displays it in several plot formats.

A TDM2 system consists of a Monitoring Rack, a Communications Processor, and a host computer that runs TDM2 Software. The Monitoring Rack is a Bently Nevada 3300 continuous monitoring rack, which has built-in dynamic and static data ports from which the Communications Processor acquires vibration data. The Communications Processor is a Bently Nevada Transient Data Manager or Transient Data Interface External, dedicated computers that acquire and temporarily store data, and communicate with the host TDM2 computer. The host computer is a PC (386 or better) running Transient Data Manager 2 Software.

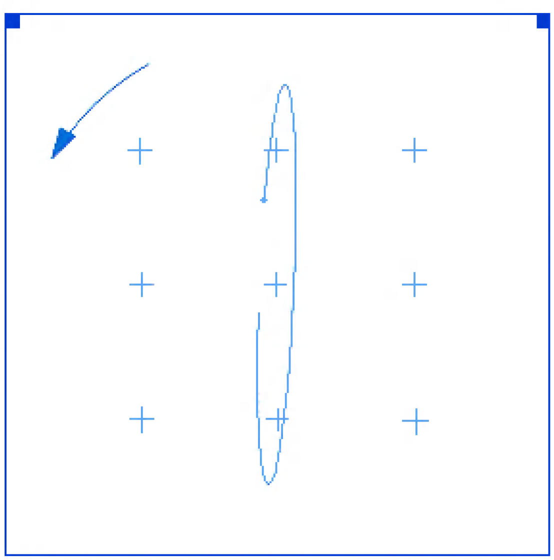

Figure 6
Compensated 1X orbit plot

TDM2 Systems are usually permanent installations. A single TDM2 host computer can track 12 monitoring racks, each with a maximum of 24 points. TDM2 computers can be networked, with other TDM2 computers, with plant and DCS computers, and with Bently Nevada's Engineer Assist™ expert system. With networking, a TDM2 System can track up to 120 Monitoring Racks of up to 24 points each. TDM2 is another of Bently Nevada's systems that "move data not people."

Like the TDM2 System, the ADRE (Automated Diagnostics for Rotating Equipment) for Windows System collects and processes data during both transient and steady state machine operation. However, ADRE for Windows is a portable system.

An ADRE for Windows System consists of a 208 DAIU, a computer interface card, and a host computer running ADRE for Windows Software. The 208 DAIU is a small, portable dedicated computer which collects, stores and processes up to 8 channels of data. Two 208 DAIUs can be connected to an ADRE for Windows computer for 16 channels of simultaneous data collection. The interface card connects the 208 DAIU to the host computer. The host computer is a PC (386 or better); it can be a desktop computer, but a notebook computer is more appropriate for this portable system.

ADRE for Windows runs under Microsoft Windows, which gives it enormous power and flexibility. You can view data from several transducers, each in its own window, view data from one transducer in several different plot formats, each in its own window, or both.

Efficiency is important in any data analysis, transient or steady state. A plant's vibration specialist is most productive when he is at his desk analyzing data, not walking a route collecting it. All Bently Nevada online diagnostic systems acquire vibration data automatically, freeing machine specialists from that repetitive task. The data is also available on another computer via a modem and phone line, so in most cases a machinery specialist need not be onsite to analyze a machine. All Bently Nevada perma-

nently-installed diagnostic systems, including Transient Data Manager® 2, support network communications. In a network, machine data is always available to operators and managers. They can use it to make better-informed operating decisions that extend machine availability and life. Bently Nevada systems communicate with each other and with plant control and DCS computers.

For more information on transient data analysis, see the references for this article. For more information on Bently Nevada's systems for transient data capture and analysis, systems with remote access and systems that network with your plant's process or DCS system, contact your nearest Bently Nevada Sales and Service representative. ■

References:

Transient data analysis is a broad subject, and space permitted us to explain only some of its applications. Bently Nevada has published many articles related to transient data analysis, in Bently Rotor Dynamics Research Corporation papers, in Bently Nevada Corporation Application Notes, and in Orbit magazine article reprints. Some of the papers are listed here. They are available to you - contact the Orbit editor, at the address, telephone or fax number listed in the front of this magazine.

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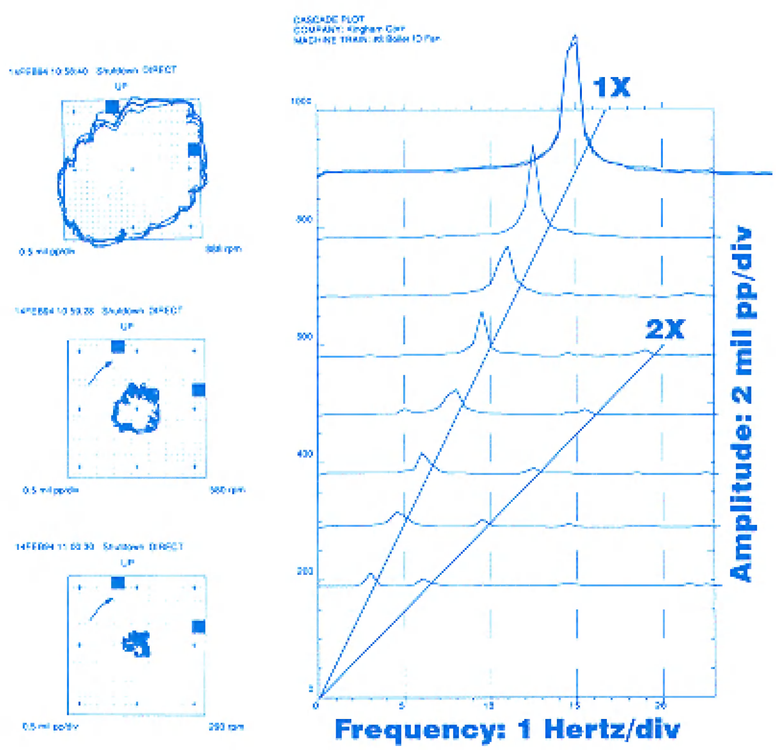

Figure 7
Spectrum cascade plot with "plus" orbit plots